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Stirling System Modeling for Space Nuclear Power Systems

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Abstract

A dynamic model of a high-power Stirling convertor has been developed for space nuclear power systems modeling. The model is based on the Component Test Power Convertor (CTPC), a 12.5-kWe free-piston Stirling convertor. The model includes the fluid heat source, the Stirling convertor, output power, and heat rejection. The Stirling convertor model includes the Stirling cycle thermodynamics, heat flow, mechanical mass-spring damper systems, and the linear alternator. The model was validated against test data.

Both nonlinear and linear versions of the model were developed. The linear version algebraically couples two separate linear dynamic models; one model of the Stirling cycle and one model of the thermal system, through the pressure factors.

Future possible uses of the Stirling system dynamic model are discussed. A pair of commercially available 1-kWe Stirling convertors is being purchased by NASA Glenn Research Center. The specifications of those convertors may eventually be incorporated into the dynamic model and analysis compared to the convertor test data. Subsequent potential testing could include integrating the convertors into a pumped liquid metal hot-end interface. This test would provide more data for comparison to the dynamic model analysis.

Introduction

Dynamic modeling and simulation play an important role in the development of space nuclear power systems. Because of the cost of prototypes, and complexity of the system, accurate models are beneficial for trade studies, evaluation of design options, dynamic performance prediction, failure effects analysis, design optimization, and controls development.

This paper describes the dynamic modeling of the Stirling convertor, which converts heat energy into electrical energy. The Stirling convertor includes thermal, mechanical, fluid, magnetic, and electrical dynamic elements. While models have been developed for the various elements of the convertor, few models combine them into one model. The model described herein contains all of these elements, allowing the study of complex system dynamic interactions among subsystems. It is a non-linear time-domain model that can simulate transient and dynamic phenomena over the entire range of operating conditions, from start-up to full power.

To facilitate systems development and integration, a linear version of the model was developed. Dynamic model parameters are calculated for the Component Test Power Convertor (CTPC) based on data from published reports (refs. 1 to 4).

Nomenclature

A_d	Displacer area (m^2)
A_p	Piston area (m^2)
A_{rod}	Displacer rod area (m^2)
C_d	Displacer damping (N·s/m)
C_h	Heater head thermal capacitance (J/K)
C_p	Piston damping (N·s/m)
C_t	Tuning capacitance (μF)
f	Operating frequency (Hz)
I_{alt}	Alternator current (A)
K_{bounce}	Bounce space spring rate (N/m)
K_d	Displacer spring rate (N/m)
K_{magnet}	Alternator magnet space spring rate (N/m)
K_p	Piston spring rate (N/m)
L_{alt}	Alternator inductance (H)
M_d	Effective displacer mass (kg)
M_p	Effective piston mass (kg)
M_W	Gas molecular weight
N	Number of turns on the alternator winding
P	Stirling cycle dynamic pressure (Pa)
P_c	Compression space pressure (Pa)
P_e	Expansion space pressure (Pa)
P_{exp}	Expansion space PV power (W)
P_d	Displacer pressure factor (Pa/m)
P_p	Piston pressure factor (Pa/m)
Q_{in}	Heat into the Stirling cycle (W)
Q_{out}	Heat rejected (W)
Q_{source}	Heat delivered by the heat source (W)
R_{alt}	Alternator electrical resistance (Ω)
R_c	Compression space thermal resistance (W/m·K)
R_e	Expansion space thermal resistance (W/m·K)
R_{gas}	Gas constant (J/kg·K)

R_h	Heater head thermal resistance (W/m·K)
$R_{hhCondLoss}$	Heater head conduction loss thermal resistance (W/m·K)
R_{ins}	Insulation thermal resistance (W/m·K)
t	time (sec)
T_{alt}	Alternator temperature (K)
$T_{ambient}$	Ambient temperature (K)
T_c	Compression space gas temperature (K)
T_e	Expansion space gas temperature (K)
T_h	Hot heat exchanger gas temperature (K)
T_k	Cold heat exchanger gas temperature (K)
T_r	Regenerator space gas temperature (K)
T_{source}	Heat source temperature (K)
V_c	Compression space gas volume (m^3)
V_{co}	Mean compression space gas volume (m^3)
V_e	Expansion space gas volume (m^3)
V_{eo}	Mean expansion space gas volume (m^3)
V_h	Hot heat exchanger gas volume (m^3)
V_k	Cold heat exchanger gas volume (m^3)
V_r	Regenerator gas volume (m^3)
X_d	Displacer position (m)
x_{damp}	Displacer position amplitude (m)
x_p	Piston position (m)
x_{pamp}	Piston position amplitude (m)
ΔP	Pressure drop across regenerator, heater and cooler (Pa)
η_{mag}	Alternator magnetic efficiency
Φ	Flux (Wb)
ϕ_d	Displacer phase angle (deg)
ϕ_p	Pressure phase angle (deg)
ω	Operating frequency (rad/sec)

The Component Test Power Convertor

The CTPC is a 12.5-kWe free-piston Stirling convertor designed, built, and tested in the late 1980s and early 1990s by Mechanical Technology Inc. (MTI) (ref. 5). The convertor took heat from radiant electric heaters or from heat pipes and converted it to about 70-Hz AC electric power. With an input temperature of 800 K and a temperature ratio of 2.0, overall conversion efficiency was about 22 percent.

This paper summarizes the modeling of the CTPC system using NASA Glenn Research Center (GRC) System Dynamic Model (SDM) (refs. 6 and 7). The SDM models the system from the heat source to the Stirling convertor to output power and rejected heat. The heat source model includes insulation loss and the temperature drops from the heat source to the Stirling convertor heater. The Stirling convertor model includes the Stirling cycle thermodynamics, heat flow, mechanical mass-spring damper systems, and the linear alternator. The heat rejection system model assumes a fixed temperature heat sink, but could be expanded to include a more realistic radiator system.

CTPC Dynamic Model

A high-level schematic of the model is shown in figure 1. Starting at the bottom of the figure, the heat input to the Stirling cycle was represented by the heat input Q_{source}, shown with two flow paths emanating from it. One flow path represents the heat lost through the insulation to ambient temperature and has the thermal resistance R_{ins}. The other flow path represents the heat conducted to the heater. There is a temperature drop from the heat source to the heater. The temperature drop was modeled by the resistance R_h. The thermal time constant of the heat input system was modeled by the thermal capacitance C_h. This parameter was determined based on the mass of the CTPC heater and its heat capacity.

Not all of the heat entering the heater of the Stirling convertor goes into the Stirling cycle. Some is conducted to the cold end through paths including the heater head wall,

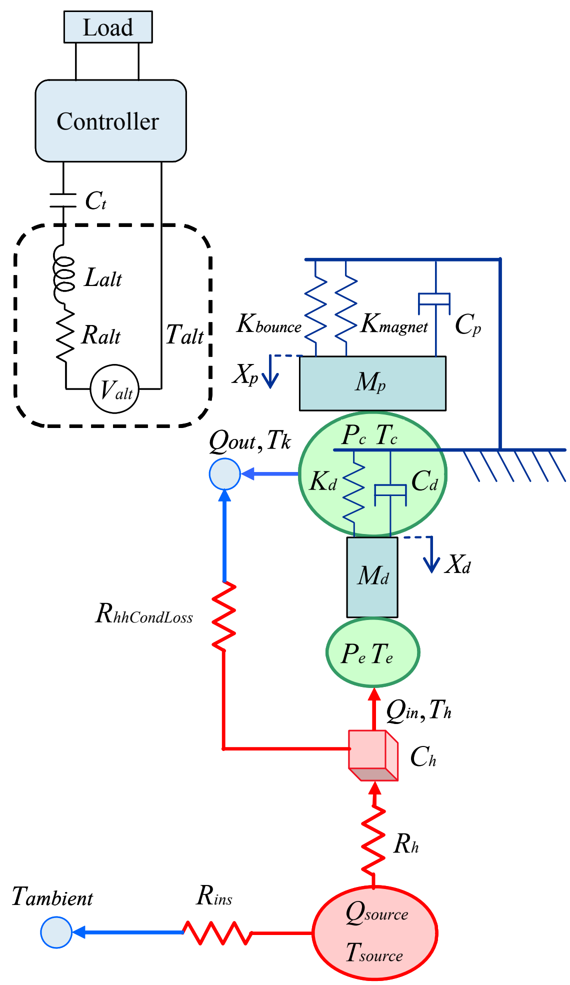

Figure 1.—Schematic of the CTPC dynamic model.

inner cylinder, helium gas, regenerator matrix, and the displacer. This loss was modeled by the heat flow through the thermal resistance $R_{hhCondLoss}$.

The Stirling cycle thermodynamics are modeled based on the Schmidt model (ref. 8), which assumes an isothermal Stirling cycle. Pumping losses through heat exchangers and the regenerator are considered. The thermodynamic portion of the model determines the internal expansion space and compression space gas pressures P_e and P_c. These pressures generate the driving forces on the displacer mass M_d and the piston mass M_p. Gas spring and damping forces on the displacer are represented by K_d and C_d. The piston is subjected to spring force K_p, the sum of the bounce space spring K_{bounce} and the magnet spring K_{magnet}.

The alternator produces a damping force on the piston based on the alternator current. Alternator electrical dynamics are determined by the alternator resistance, inductance, motor constant, and the tuning capacitor.

The Stirling convertor was assumed to be rigidly connected to the ground. Casing motion, dual-opposed dynamics, and the effect of a dynamic balancer could be added to this model if desired.

The model shown in figure 1 captures the major dynamics of the CTPC system. These include the thermal, mechanical, electrical, controller, fluid, and gas dynamics. With these characteristics captured in one system model, dynamic interactions and effects can be studied and analyzed.

Linear Dynamic Stirling Model

The ultimate selection of a model configuration is determined by a number of factors, including:

- Dynamics required based on simulation objectives
- Simulation time
- Model platform

In order to provide a model that captures the necessary Stirling convertor dynamics but does not require excessive simulation time and can be easily ported to other platforms (e.g., Matlab/Simulink) (The MathWorks, Inc.), a linear model of the CTPC was developed based on the nonlinear SDM model (ref. 9). The nonlinear model captured physics-based details of the CTPC and determined linearized parameters. These linearized parameters can be used to create a model in Matlab/Simulink or any other dynamic modeling tool that can be integrated into a larger system.

Linear dynamic models have been used for many years to design and analyze free-piston Stirling engines. These models show the piston and displacer masses are acted upon by spring forces and damping forces. The spring forces can include both mechanical components and components produced by the Stirling cycle pressure wave. The alternator is considered rigidly coupled to the piston, and the electrical load is assumed to be purely resistive. The equations for this model are as follows:

$$\frac{dx_d}{dt} = \dot{x}_d \tag{1}$$

$$\begin{aligned}\frac{d\dot{x}_d}{dt} = &-\left(K_d + A_r\frac{\partial P}{\partial x_d}\right)\frac{1}{M_d}x_d \\ &+\left(A_d\frac{\partial \Delta P}{\partial \dot{x}_d} - C_d\right)\frac{1}{M_d}\dot{x}_d \\ &- A_r\frac{\partial P}{\partial x_p}\frac{1}{M_d}x_p + \left(A_d\frac{\partial \Delta P}{\partial \dot{x}_p}\right)\frac{1}{M_d}\dot{x}_p\end{aligned} \tag{2}$$

$$\frac{dx_p}{dt} = \dot{x}_p \tag{3}$$

$$\begin{aligned}\frac{d\dot{x}_p}{dt} = &-A_p\frac{\partial P}{\partial x_d}\frac{1}{M_p}x_d \\ &-\left(K_p + A_p\frac{\partial P}{\partial x_p}\right)\frac{1}{M_p}x_p \\ &-\frac{C_p}{M_p}\dot{x}_p + N\frac{d\Phi}{dx_p}\frac{1}{\eta_{mag}}\frac{1}{M_p}I_{alt}\end{aligned} \tag{4}$$

Equations (1) through (4) represent the dynamic equations for the displacer and the piston. In equation (2), the force due to the pressure wave's action on the displacer area A_d has been accounted for in two parts. This is because the pressure is expressed as a linear combination of the piston and displacer positions. In order to reconstruct the pressure wave, the contribution of both components must be considered. The same discussion applies to the pressure force on the piston in equation (4).

Thermal System Model

Figure 2 shows the thermal system modeled as a series of thermal resistances with a heat capacitance, heat input, and temperature sinks. It is analogous to figure 1 with the electromechanical components removed. The schematic shows the system as it is configured in Ansoft Simplorer (Ansoft Corporation) the platform used for the CTPC model.

Thermal energy from the heater Q_{source} is divided between two flow paths: (1) heat lost through the insulation to ambient is determined from R_{ins} and (2) heat transferred to the heater head is determined from R_h. The resulting heat flow to the heater head causes an increased convertor hot-end temperature T_h.

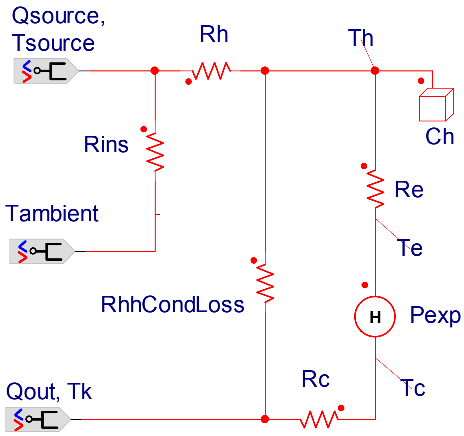

Figure 2.—Schematic of the thermal system model.

Thermal energy from T_h is divided among three flow paths: (1) heat lost via conduction through the convertor housing and displacer is determined from $R_{hhCondLoss}$, (2) heat transferred to the gas in the expansion space is determined from the gas dynamics, and (3) heat transferred to the heater head material is determined from the heater head thermal capacitance C_h. Thermal capacitance only becomes apparent during transient analysis; at steady-state operation, the heater head material temperature is constant, and heat flow is divided between the conduction loss and heating of the gas expansion space, with the bulk of the heat being transferred to the latter.

Heat is rejected from the convertor to the cold sink temperature T_k. Most of the rejected heat flows from the working fluid. The remainder of the heat rejected is from the conduction losses.

The thermal model determines the hot and cold gas temperatures (T_h and T_k) for the Stirling cycle, but does not calculate cycle performance or convert heat energy into mechanical energy. This is accomplished by coupling the linear Stirling model to the thermal system model. The next section will further discuss how the model coupling is performed.

Coupling Stirling Linear Model and Thermal System Model

To complete the linear model of the CTPC, the Stirling linear model has to be coupled to the thermal system model. Because the response time of the Stirling cycle is on the order of milliseconds, while the response time of the thermal system is on the order of seconds or minutes, the two systems were algebraically coupled together through the pressure factors.

The pressure factors are the partial derivatives of the pressure wave P with respect to displacer position x_d and piston position x_p. The equations to couple the systems can be derived from the Stirling cycle equations as derived by Urieli and Berchowitz (ref. 8).

$$P = M_w \bullet R_{gas} \bullet \left(\frac{V_h}{T_h} + \frac{V_r}{T_r} + \frac{V_k}{T_k} + \frac{V_e}{T_e} + \frac{V_c}{T_c} \right)^{-1} \tag{6}$$

In equation (6) the volumes V_e and V_c are functions of x_d and x_p. The other variables are assumed to be fixed over a given cycle.

$$V_e = V_{eo} - A_d x_d \tag{7}$$

$$V_c = V_{co} - A_p x_p + (A_d - A_{rod}) x_d \tag{8}$$

The temperatures T_c and T_e can be expressed in terms of the heat into the heater head Q_h based on the thermal circuit shown in figure 2, where the thermal resistance R_e is used to calculate the temperature drop from T_h to T_e, and R_c is used to calculate the temperature drop from T_c to T_k.

$$T_e = T_h - Q_h R_e \tag{9}$$

$$T_c = Q_h R_c + T_k \tag{10}$$

Also, the regenerator temperature can be expressed as the log mean temperature between T_h and T_k:

$$T_r = \frac{T_h - T_k}{\ln(T_h / T_k)} \tag{11}$$

Substituting equations (7) through (10) into equation (6) and differentiating yields the following expressions for the pressure factors:

$$\begin{aligned} \frac{\partial P}{\partial x_p} = {} & \frac{A_p M_w R_{gas}}{(Q_h R_c + T_k)} \\ & \bullet \left[\frac{V_{co} - A_p X_p + (A_d - A_{rod}) X_d}{Q_h R_c + T_k} + \frac{V_k}{T_k} \right. \\ & \left. + \frac{V_r \ln(T_h / T_k)}{T_h - T_k} + \frac{V_h}{T_h} + \frac{V_{eo} - A_d X_d}{T_h - Q_h R_e} \right]^{-2} \end{aligned} \tag{12}$$

$$\begin{aligned} \frac{\partial P}{\partial x_d} = {} & -M_w R_{gas} \left[\frac{(A_d - A_{rod})}{Q_h R_c + T_k} - \frac{A_d}{(T_h - Q_h R_e)} \right] \\ & \bullet \left[\frac{V_{co} - A_p X_p + (A_d - A_{rod}) X_d}{Q_h R_c + T_k} + \frac{V_k}{T_k} \right. \\ & \left. + \frac{V_r \ln(T_h / T_k)}{T_h - T_k} + \frac{V_h}{T_h} + \frac{V_{eo} - A_d X_d}{T_h - Q_h R_e} \right]^{-2} \end{aligned} \tag{13}$$

The pressure factors are now functions of T_h, T_k, Q_h, which can be determined based on the dynamics of the thermal system. The pressure factors are also functions of x_p and x_d.

To algebraically couple the thermal system with the Stirling model, it is also necessary to calculate the expansion space PV power P_{exp}. The expansion space PV power is approximately equal to the heat flow into the Stirling cycle:

$$Q_h \approx f \oint P dV \tag{14}$$

For the expansion space,

$$V_e = V_{eo} - A_d x_d \tag{15}$$

Since

$$x_d(t) = x_{d_{amp}} \sin(\omega t + \phi_d) \tag{16}$$

$$x_p(t) = x_{p_{amp}} \sin(\omega t) \tag{17}$$

then

$$\frac{\mathrm{d}V_e}{dt} = -A_d x_{d_{amp}} \omega \cos(\omega t + \phi_d) \tag{18}$$

Also, the pressure in the expansion space is a function of the pressure factors.

$$P(t) = P_d x_d(t) + P_p x_p(t) \tag{19}$$

Substituting equations (18) and (19) into equation (14) and integrating, then multiplying by the frequency f to give the expansion space PV power:

$$P_{\exp} = \pi A_d f x_{d_{amp}} x_{p_{amp}} P_p \sin \phi_d \tag{20}$$

Simulation Results

The CTPC was modeled in SDM based on parameters available in a NASA document "Stirling Space Engine Program, Volume 1—Final Report." (ref. 1) Key model parameters are provided in the table A–I in appendix A. A comparison of CTPC test data, SDM simulation, and linearized model simulation are summarized in table I. The SDM model shows good correlation between the test data for most parameters, with the larger differences attributable to the isothermal model assumption used in SDM. This assumption results in a reduced pressure amplitude but higher pressure phase angle. Further refinements to the SDM model of the CTPC could be made if necessary to improve model fidelity, including enhancement of the thermodynamic model (ref. 10).

The linearized CTPC model tracks well to the nonlinear model for most parameters.

TABLE I.—CTPC MODEL SIMULATION RESULTS VERSUS TEST DATA

			SDM nonlinear model		Linearized model	
Parameter	**units**	**800 K test data**	Value	error vs. test data	Value	error vs. nonlinear model
Power out	W	12,780	12,891	0.9%	12,863	-0.2%
current	Arms		48.09		48.04	-0.1%
voltage	Vrms		401.0		402.3	0.3%
frequency	Hz	67.45	67.48	0.0%	66.77	-1.1%
XDamplitude	m	0.01480	0.01266	-14.4%	0.01294	2.2%
XPamplitude	m	0.01344	0.01363	1.4%	0.01372	0.7%
displacer phase angle	deg	70.83	68.87	-2.0	81.53	12.7°
mean pressure	Pa	15,000,000	15,020,114	0.1%	n/a	
pressure amplitude	Pa	1,600,000	1,247,179	-22.1%	n/a	
pressure phase angle	deg	-12.48	-15.79	-3.3	n/a	
alternator efficiency	%	87.84%	90.62%	3.2%	n/a	
Th	K	800	799.7	0.0%	799.7	0.0%
Te	K	776	779.7	0.5%	779.6	0.0%
Tc	K	418.5	415.4	-0.8%	415.5	0.0%
Tk	K	400	400.0	0.0%	400.0	0.0%

Future Modeling Efforts

The CTPC was the first benchmark for the SDM against a multi-kilowatt convertor. However, the CTPC represents hardware that was developed in the late 1980s for which there is limited available test data. NASA GRC is in the process of procuring dual-opposed 1-kW Stirling convertors with the intent to investigate heater head concepts. The convertor will not be flight-like, but does represent state-of-the-art hardware.

The new hardware provides the opportunity to compare the SDM analysis to test data from another high-powered Stirling convertor. The convertor specifications will be incorporated into the SDM to generate both linear and non-linear models with the goal of simulating dynamic performance. The convertor will initially be tested at GRC with an electric cartridge heater as the heat source. This test will provide data for comparison to analytical predictions generated by the SDM. Because the SDM emphasizes convertor electromechanical behavior, it would be desirable to obtain data that involves changes in load and piston amplitude and/or frequency. There is potential for tests that include various controller set points to capture the off-nominal electromechanical behavior of the convertor in addition to its steady-state design operating point.

A power conversion system for space applications might consist of a set of Stirling convertors coupled to a liquid metal-cooled nuclear fission reactor. Therefore, the test convertors could be modified at the header head to permit operation with a liquid metal heat transfer loop such as the NaK-cooled reactor simulator recently installed at NASA Marshall Space Flight Center (MSFC) (ref. 11). This test would provide the opportunity to enhance the hot-end thermal modeling in the SDM and supply additional convertor test data.

Conclusions

Dynamic modeling of high powered Stirling convertors facilitates for the development of space nuclear power systems. An accurate dynamic model that comprises all aspects of the convertor—from the heat source to heat rejection to electromechanical power generation—can assist in the design of an optimized system. Developing such a model requires a firm understanding of the Stirling convertor subsystems, as well as methods for validating the model's accuracy.

This paper presented the model configuration and methodology employed to simulate a free-piston Stirling convertor. The model was designed to represent the hardware for the multi-kilowatt CTPC, and results were compared to the limited test data available. Discrepancies between the simulation results and data are attributed to the model assumptions. Further refinement of the model is possible in order to achieve a higher fidelity model.

Additional simulations of multi-kilowatt convertors will take place as test data and hardware specifications become available. The more opportunities one has to validate the model, the more valuable it will be for the successful design of a dynamic power conversion system.

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Appendix A

TABLE A–I.—CTPC MODEL PARAMETERS

Parameter	Value	Units	Description
Cd	42.7	N·s/m	Displacer damping
Cp	40.1	N·s/m	Piston damping
Md	2.17	kg	Effective displacer moving mass
Mp	13.176	kg	Effective piston moving mass
DiaD	0.1143	m	Displacer diameter
Odp	0.13716	m	Piston diameter
OdRod	0.0244835	m	Displacer rod diameter
Vco	1.2091e-3	m^3	Mean compression space volume
Veo	4.279e-4	m^3	Mean expansion space volume
Vbounce	0.010194	m^3	Bounce space volume
VdispAft	8.7072e-4	m^3	Aft displacer gas spring volume
AdispAft	3.7865e-3	m^2	Aft displacer gas spring area
VdispForward	7.9509e-3	m^3	Forward displacer gas spring volume
AdispForward	4.2573e-3	m^2	Forward displacer gas spring area
Vh	1.1421e-4	m^3	Heater volume
HHxNchan	1900		Heater number of channels
HHxDia	1.016e-3	m	Heater hole diameter
HHxLg	0.05969	m	Heater Length
Vr	8.2183e-4	m^3	Regenerator volume
RgnLg	0.0376	m	Regenerator Length
RgnOD	0.22780	m	Regenerator OD
RgnID	0.11690	m	Regenerator ID
rhoR	0.728		Regenerator fill factor (porosity)
Rdw	50.8	µm	Diameter of regenerator fiber
Vk	1.73844e-4	m^3	Cooler volume
CHxNchan	2580		Cooler number of channels
CHxLg	0.07493	m	Cooler Length
CHxChW	5.334e-4	m	Cooler channel width
CHxChL	1.464e-3	m	Cooler channel length
Rgas	2077	J/(kg·K)	Gas constant for helium
gammaHe	1.6		Ratio of specific heats C_p/C_v
Ralt	0.14203	Ω	Alternator resistance at operating temperature
Lalt	0.0145	H	Stator Self Inductance
Ke	68.3	V·sec/m	Alternator voltage constant
Ct	346	µF	Tuning capacitor
Ch	2150	J/K	Heater thermal inertia
R_e	0.000585	K/W	Thermal resistance to model temperature drop between T_h and T_e
R_c	0.000451	K/W	Thermal resistance to model temperature drop between T_c and T_k

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14. ABSTRACT A dynamic model of a high-power Stirling convertor has been developed for space nuclear power systems modeling. The model is based on the Component Test Power Convertor (CTPC), a 12.5-kWe free-piston Stirling convertor. The model includes the fluid heat source, the Stirling convertor, output power, and heat rejection. The Stirling convertor model includes the Stirling cycle thermodynamics, heat flow, mechanical mass-spring damper systems, and the linear alternator. The model was validated against test data. Both nonlinear and linear versions of the model were developed. The linear version algebraically couples two separate linear dynamic models; one model of the Stirling cycle and one model of the thermal system, through the pressure factors. Future possible uses of the Stirling system dynamic model are discussed. A pair of commercially available 1-kWe Stirling convertors is being purchased by NASA Glenn Research Center. The specifications of those convertors may eventually be incorporated into the dynamic model and analysis compared to the convertor test data. Subsequent potential testing could include integrating the convertors into a pumped liquid metal hot-end interface. This test would provide more data for comparison to the dynamic model analysis.					
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